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1 [0007] In a separate development of heat exchangers, evaporative cooling may be used for
2 cooling air below its wet bulb temperature and up to its dew point temperature in an improved
3 cooling cycle. This Maisotsenko Cycle is disclosed in various configurations in various
4 patents, including U.S. Patents 5,453,223; 6,497,107; and 6,581,402. Additional information
5 for this cycle is available on the worldwide web at idalextechnologies.com.

6 [0008] Accordingly, it is desired to improve heat exchanger performance in turbine cycles,
7 for example, employing yet another advancement in the Maisotsenko Cycle.

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BRIEF SUMMARY OF THE INVENTION

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11 [0009] A duplex exchanger includes first and second heat exchangers each including a main
12 flow channel and a cooperating counterheat channel. The first counterheat channel is joined
13 to the first main flow channel for receiving a cooled primary stream therefrom. The second
14 counterheat channel is also joined to the first main channel splitting the primary stream
15 therefrom. An evaporative coolant is injected into the first counterheat channel, and an
16 evaporative saturant is injected into the second counterheat channel. Heat from the initially
17 hot primary stream in the first exchanger evaporates the coolant in the first counterheat
18 channel for self-cooling the primary stream in the first main channel. Heat from a hot
19 secondary stream channeled through the second main channel evaporates the saturant in the
20 second counterheat channel for adding mass to the primary stream channeled therethrough.

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BRIEF DESCRIPTION OF THE DRAWINGS

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24 [0010] The invention, in accordance with preferred and exemplary embodiments, together
25 with further objects and advantages thereof, is more particularly described in the following
26 detailed description taken in conjunction with the accompanying drawings in which:

27 [0011] Figure 1 is a schematic representation of a split counterflow, duplex exchanger
28 saturator in an exemplary turbine cycle application.

29 [0012] Figure 2 is a schematic representation of the duplex exchanger saturator of Figure 1
30 in accordance with an exemplary tube-in-shell configuration for high pressure applications.

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1 [0013] Figure 3 is a transverse sectional view through the lower heat exchanger illustrated in
2 Figure 2 and taken along line 3-3.

3 [0014] Figure 4 is a isometric view of an exemplary finned tube in the dual heat exchangers
4 of Figure 2.

5 [0015] Figure 5 is a schematic representation of single shell duplex exchanger saturator
6 including dual heat exchangers in an alternate embodiment.

7 [0016] Figure 6 is a partly sectional, schematic view of a tube-in-plate configuration of the
8 saturator illustrated in Figure 1 in accordance with another embodiment.

9 [0017] Figure 7 is an elevational sectional view of a portion of the saturator illustrated in
10 Figure 6 and taken along line 7-7.

11 [0018] Figure 8 is a partly exploded and sectional view of a corrugated plate duplex
12 exchanger saturator corresponding with Figure 1 in accordance with another embodiment.

13 [0019] Figure 9 is a transverse sectional view through a portion of the plate saturator
14 illustrated in Figure 8 and taken along line 9-9.

15 [0020] Figure 10 is another transverse sectional view through a portion of the plate saturator
16 illustrated in Figure 8 and taken along line 10-10.

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18 DETAILED DESCRIPTION OF THE INVENTION

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20 [0021] Illustrated schematically in Figure 1 is a gas turbine system 10 including in serial
21 flow communication a multistage compressor 12, combustor 14, and turbine 16 which may
22 have any conventional configuration. Ambient air 18 is pressurized in the compressor during
23 operation and mixed with fuel in the combustor for generating hot combustion gases 20 which
24 flow downstream to the turbine.

25 [0022] Energy is extracted from the combustion gases in turbine rotor blades which are
26 supported to a rotor disk joined by a shaft to the rotor blades of the compressor for providing
27 power thereto. Output power may be obtained from the high pressure turbine, or in another
28 conventional configuration may be obtained from a low pressure turbine disposed downstream
29 therefrom and mounted on a separate driveshaft (not shown). The output power may be used
30 for any suitable purpose, such as powering an electrical generator (not shown) in a typical

1 power generation system.

2 [0023] As indicated above, the efficiency of the turbine system illustrated in Figure 1 is
3 based in part on the efficiency of compression of the ambient air 18, efficiency of combustion
4 of the air and fuel in the combustor, and in additional part on the efficiency of energy
5 extraction from the combustion gases in the turbine. One manner of improving the overall
6 efficiency of the turbine system illustrated in Figure 1 is to add mass to the pressurized air 18
7 discharged from the compressor for increasing the mass flowrate of the combustion gases 20
8 channeled through the turbine.

9 [0024] Shown schematically in Figure 1 is an evaporative duplex counterheat exchanger or
10 apparatus 22 configured as a humidifier or saturator for cooperating with the components of
11 the turbine system for efficiently introducing additional mass to the compressor discharge air
12 18 during operation. More specifically, the duplex exchanger is an assembly of components
13 including a first or primary heat exchanger 24 and a cooperating second or secondary heat
14 exchanger 26.

15 [0025] The first heat exchanger includes a first main flow channel 28 for receiving a primary
16 gas stream in the exemplary form of the hot compressed air 18 from the compressor suitably
17 joined thereto. The first exchanger also includes a first counterheat channel 30 joined in flow
18 communication with the first main channel 28.

19 [0026] Correspondingly, the second heat exchanger 26 includes a second main flow channel
20 32 for receiving a secondary fluid stream in the exemplary form of the hot combustion gases
21 20 discharged from the turbine 16 suitably joined thereto. The second heat exchanger also
22 includes a second counterheat channel 34 joined in flow communication with the first heat
23 exchanger.

24 [0027] The first and second counterheat channels may be configured in any conventional
25 manner to channel flow in opposition, or countercurrent, to the main flows being channeled in
26 the corresponding first and second main flow channels for exchanging heat therewith.
27 Accordingly, the counterheat is typically effected by conventionally introducing counterflow
28 in any suitable direction opposite to the direction of the main flow, and is typically oblique
29 thereto in serpentine fashion, with a corresponding amount of lateral crossflow. In other
30 embodiments, counterheat may be introduced by lateral crossflow alone, or other forms of

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1 countercurrent flow.

2 **[0028]** Means 36 are provided in the duplex exchanger for injecting an evaporative fluid 38
3 into the first counterheat channel 30 during operation. The injecting means may have any
4 conventional form for suitably injecting or dispensing or spraying or wicking or otherwise
5 introducing the evaporative fluid into the counterheat channel for cooperating with the
6 primary gas stream therein as further described hereinbelow. In the exemplary application
7 illustrated in Figure 1, mass may be added to the compressed air 18 by the introduction of
8 water vapor therein, and therefore the evaporative fluid 38 is simply water, injected in any
9 convenient manner for this purpose.

10 **[0029]** However, the duplex exchanger may be used in various other applications in which
11 cooling of a fluid is desired, or saturation of one fluid into another is desired in various
12 amounts, and therefore the evaporative fluid may have any suitable form, including petroleum
13 liquids or diesel fuel for specific examples associated with turbine cycles. In other
14 applications, other evaporative fluids may be used.

15 **[0030]** The two heat exchangers cooperate with each other by suitably joining the second
16 counterheat channel 34 in flow communication with the discharge end of the first main
17 channel 28. In this way, the initially hot primary stream 18 is initially self-cooled in the first
18 main channel 28 and then delivered to the inlet end of the second counterheat channel 34 in
19 the second heat exchanger.

20 **[0031]** The injecting means 36 are also provided for injecting a common evaporative fluid or
21 saturant, also designated 38, into the second counterheat channel 34 in a manner similar to
22 that for injecting the evaporative fluid or coolant into the first counterheat channel. Since the
23 same primary stream 18 is being channeled through both counterheat channels 30,34 the same
24 evaporative fluid 38 may be used in both heat exchangers for similar purposes in providing
25 evaporative cooling of the primary stream in stages, and the humidification or saturation
26 thereof also in stages for reaching full saturation in the primary stream in the preferred
27 embodiment.

28 **[0032]** The two heat exchangers are similar in configuration and operation including both
29 main and counterheat channels for the primary and secondary flow streams channeled
30 therethrough. The primary stream 18 is split at the discharge end of the first main flow

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1 channel 28, with a first, and preferably major portion being channeled to the inlet end of the
2 second counterheat channel 34; and a second, and preferably minor portion of the split
3 primary stream being channeled to the inlet end of the first counterheat channel 30. The split
4 primary streams then flow in counterheat over the two main flow channels 28,32 to the
5 opposite ends thereof.

6 [0033] In a preferred embodiment, the second counterheat channel 34 receives its stream
7 from the discharge end of the first main channel 28, and is additionally joined in flow
8 communication with the discharge end of the first counterheat channel 30 for receiving the
9 split second portion of the primary stream for rejoinder with the split first portion of the
10 primary stream near the downstream end of the second counterheat channel 34. The rejoined
11 primary stream is then discharged from the second heat exchanger 26 through a suitable outlet
12 thereof joined in flow communication with the combustor 14 of the turbine system.

13 [0034] The duplex exchanger 22 illustrated schematically in Figure 1 may be practiced in
14 various configurations using conventional forms of common heat exchangers. For example,
15 Figure 2 illustrates a first embodiment of the duplex exchanger configured for use as a split
16 counterflow saturator 22 in a tube-in-shell configuration for use with the high pressure
17 compressor discharge air 18 which may be on the order of 50 atmospheres. Figure 2 also
18 illustrates in flowchart form an exemplary method of using the duplex exchanger saturator 22
19 for saturating the hot primary gas stream 18 with the evaporative fluid 38 in the exemplary
20 form of water.

21 [0035] The primary gas stream 18 is the pressurized discharge air from the compressor 12 in
22 the exemplary turbine system, which is relatively hot due to compression therein. The
23 primary stream may have other forms of conventional gases such as natural gas or nitrogen for
24 the exemplary turbine system illustrated; or may have any other chemical composition for
25 which saturation thereof with secondary fluid is desired.

26 [0036] Accordingly, the hot primary gas stream 18 is initially channeled through the first
27 main channel 28 of the first heat exchanger for being cooled therein. At the discharge end of
28 the main channel 28, the primary stream is split into two portions for flow to the
29 corresponding first and second counterheat channels 30,34.

30 [0037] The combustion gases 20 generated in the combustor 14 after their flow through the

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1 turbine 16 still have significant heat and may be used to advantage in the saturator 22.
2 Accordingly, the relatively hot secondary fluid stream 20 is suitably channeled from the
3 discharge end of the turbine 16 to the inlet end of the second main channel 32 for use in
4 heating the cooled primary stream being channeled through the second counterheat channel
5 34.

6 [0038] In typical fashion, the first main channel 28 adjoins in parallel flow the first
7 counterheat channel 30 in opposite directions, i.e., counterflow channel 30. And, the second
8 main channel 32 adjoins in parallel flow the second counterheat channel 34 in opposite
9 directions, i.e., counterflow channel 34. The hot streams 18,20 being channeled through the
10 two main channels are evaporatively cooled by the split primary stream being channeled
11 through the two counterflow channels 30,34, which correspondingly increases both the
12 temperature and humidity thereof.

13 [0039] The evaporative coolant 38 is suitably injected into the first counterflow channel 30
14 for being evaporated by the hot primary stream flowing through the cooperating first main
15 channel 28. Accordingly, as the coolant evaporates in the counterflow channel, it
16 correspondingly cools the primary stream inside the main flow channel. And, the evaporating
17 coolant adds humidity to the cooled primary stream being discharged through the first
18 counterflow channel for commencing the saturation thereof.

19 [0040] The same evaporative fluid 38, or saturant, is similarly injected into the second
20 counterflow channel 34 for being evaporated by the hot secondary stream 20 flowing through
21 the second main channel 32. The hot secondary stream 20 therefore heats the precooled
22 primary stream in the second counterflow channel 34, with the evaporation of the saturant 38
23 being used for further humidifying or saturating the primary stream 18 being heated and
24 discharged through the second counterflow channel.

25 [0041] The same primary stream 18 is preferably split at the discharge end of the first main
26 channel 28 for efficient use in both counterflow channels 30,34 of the two heat exchangers.
27 For example, about one third of the total flowrate of the primary stream may be channeled
28 back through the first counterflow channel 30, and the remaining two thirds of the primary
29 stream may be channeled back through the second counterflow channel 34.

30 [0042] The two split primary streams being channeled separately through the two

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1 counterflow channels 30,34 will be heated differently and humidified differently but may be
2 rejoined together at a suitable location in the second heat exchanger. For example, the split
3 primary stream 18 from the first counterflow channel 30 may be discharged from an outlet
4 end thereof into an intermediate stage or section of the second counterflow channel 34 to
5 generally match the temperature and humidity of the separately channeled split primary
6 stream.

7 [0043] In the lower or first heat exchanger 24 illustrated in Figure 2, compressed air 18
8 enters the tube-side main channel at one end and passes to the opposite end thereof while
9 being cooled sensibly by the evaporation of water being added to the outside of the tubes in
10 the first counterflow channel. The first portion of the cooled compressed air discharged from
11 the first main channel is released into the upper or second heat exchanger 26 and is directed
12 across the tubes of the second main channel in counterflow. The psychometric analysis shows
13 that the temperature of the air in the first heat exchanger is cooled sensibly and incrementally
14 over the length of the main channel therein towards the dew point temperature of the
15 compressed air.

16 [0044] Ideally, the compressed air in the first heat exchanger is sensibly cooled for lowering
17 the wet bulb temperature thereof to the dew point temperature of the incoming compressed air.
18 When the second portion of the cooled air exits the first main channel and is exposed to the
19 water injection, it will be humidified to the wet bulb temperature of the previously cooled dry
20 air. And, as the air continues to travel across the outside of the first main channel in
21 counterflow thereto it is heated by the air inside the main channel for raising the dew point
22 temperature of the counterflow air for ensuring saturation thereof.

23 [0045] The upper or second heat exchanger illustrated in Figure 2 receives two air streams
24 from the lower heat exchanger: one airstream at the inlet end of the second counterflow
25 channel and another airstream at an intermediate stage downstream therefrom near the
26 forward end of the heat exchanger. The first portion of the dry cooled airstream that leaves
27 the discharge end of the first main channel 28 moves to the upper heat exchanger and is
28 directed in counterflow to the hot turbine exhaust gas being channeled through the second
29 main channel 32.

30 [0046] The cool compressed air in the second counterflow channel is injected with water for

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1 humidifying or saturating the air in a manner similar to that in the first heat exchanger. The
2 hot exhaust gases in the second main channel 32 are correspondingly cooled toward the wet
3 bulb temperature of the air in the second counterflow channel 34.

4 [0047] The web bulb temperature of the air in the second counterflow channel increases as it
5 moves in counterflow to the second main flow channel 32 absorbing heat from the secondary
6 stream 20 and raising humidity of the counterflow air. At a predetermined location in the
7 second heat exchanger, the second airstream from the first counterflow channel 30 rejoins the
8 first airstream in the second counterflow channel, and together the rejoined primary stream
9 moves in counterflow to the hot turbine exhaust gases being channeled through the second
10 main channel 32.

11 [0048] Accordingly, the primary stream 18 may be evaporatively cooled in the first main
12 channel 28 for lowering the wet bulb temperature thereof toward the dew point temperature of
13 the incoming primary stream. And, the primary stream in the first counterflow channel 30
14 may be humidified or saturated to the wet bulb temperature of the cooled primary stream
15 being discharged from the first main channel 28.

16 [0049] Correspondingly, the cooled primary stream in the first counterflow channel 30 is
17 heated by the hot primary stream in the first main channel 28 for raising the dew point
18 temperature thereof and saturating the primary stream therein.

19 [0050] In the second heat exchanger, the evaporative fluid is evaporated in the primary
20 stream in the second counterflow channel 34 and thereby cools the secondary stream 20 in the
21 second main channel toward the wet bulb temperature of the primary stream in the second
22 counterflow channel.

23 [0051] As the cooled primary stream is channeled through the second counterflow channel
24 34, it is heated by the hot secondary stream in the second main channel 32 for increasing the
25 wet bulb temperature thereof as well as increasing the humidity or saturation thereof. The
26 primary stream may therefore be humidified or saturated as desired for increasing the effective
27 mass thereof and suitably discharged from the saturator for return to the combustor 14 to
28 undergo combustion therein. The additional mass of the primary airstream increases the mass
29 flowrate of the combustion gases 20 being channeled through the turbine 16 for increasing
30 efficiency of the turbine system.

1 [0052] As indicated above, the fluid injecting means 36 illustrated in Figures 1 and 2 may
2 have any suitable configuration, including various conduits, nozzles, and valves for
3 controlling the location and flowrate of the evaporative fluid being injected into both heat
4 exchangers. In one embodiment, the evaporative fluid may be injected solely along an initial
5 stage or length of the second counterflow channel 34, with the remaining terminal stage of the
6 second counterflow channel being without fluid injection yet still being subject to heating
7 from the secondary stream having its hottest temperature over the initial length of the second
8 main channel 32. The heat of the secondary stream may therefore be used for adding heat or
9 superheating the primary stream 18 in the second counterflow channel 34 downstream of the
10 initial stage thereof.

11 [0053] Similarly, the evaporative fluid may be injected solely along an initial stage of the
12 first counterflow channel 30 for superheating the split primary stream along the outlet end of
13 the first counterflow channel using the hot primary stream flowing through the first main flow
14 channel.

15 [0054] One embodiment of the duplex exchanger saturator 22 specifically configured for a
16 high pressure primary stream is illustrated schematically in Figures 2-4. The first heat
17 exchanger 24 includes a bank of first tubes 28 which collectively define the first main channel,
18 with the tubes being supported in a plurality of alternating first baffles 30 which collectively
19 define the first counterflow channel around the first tubes in the lower shell.

20 [0055] Similarly, the second heat exchanger 26 includes a bank of second tubes 32
21 collectively defining the second main channel. The second tubes are supported in a plurality
22 of alternating second baffles 34 which collectively define the second counterflow channel
23 around the second tubes in the upper shell.

24 [0056] Tube-and-baffle heat exchangers are notoriously well known in various
25 configurations, and may be combined in the manner disclosed above and modified for
26 enjoying the improved performance described herein.

27 [0057] In the exemplary embodiment illustrated in Figure 2, the first heat exchanger 24 is
28 disposed inside a first tubular shell 40, and the second heat exchanger 26 is similarly disposed
29 in another or second tubular shell 40 of similar design spaced from the first shell. The two
30 shells provide a double septum therebetween which correspondingly separates or divides the

1 two heat exchangers 24,26 from each other for controlling the desired flow streams
2 therebetween. For example, suitable conduits are provided between the two shells for
3 separately channeling the first dry-cool primary stream split and the second wet-cool primary
4 stream split from the first heat exchanger to the second heat exchanger as described above.

5 **[0058]** As illustrated in Figure 4, the first tubes 28 preferably include external grooves or
6 fins 42 for increasing wetting and heat transfer surface area thereof for the evaporative fluid.
7 The corresponding tubes 32 of the second heat exchanger may be identical to the first tubes 28
8 of the first heat exchanger and similarly include the external fins therearound. In this way,
9 efficiency is increased for evaporating the fluid around the two sets of main tubes 28,32 in the
10 two heat exchangers.

11 **[0059]** In view of the two shell design illustrated in Figure 2, the first heat exchanger 24
12 includes a first inlet 44 at a forward end thereof for receiving and providing the primary
13 stream 18 to the first tubes 28. The first exchanger also includes a first outlet 46 at an opposite
14 aft end for discharging the first split portion of the primary stream 18 from the first tubes. A
15 second outlet 48 is disposed between the inlet and first outlet of the first heat exchanger for
16 discharging the second split portion of the primary stream from the first counterflow channel
17 30.

18 **[0060]** Correspondingly, the second heat exchanger 26 includes a respective first inlet 44 at
19 its forward end for receiving and providing the secondary stream 20 to the second tubes 32. A
20 respective first outlet 46 is provided at the aft end of the first heat exchanger for discharging
21 the secondary stream 20 from the second tubes 32 as relatively cool waste fluid.

22 **[0061]** A respective second outlet 48 is also disposed in the second heat exchanger adjacent
23 the first inlet 44 thereof for discharging the saturated primary stream 18 from the second
24 counterflow channel 34 back to the combustor of the exemplary turbine system illustrated in
25 Figure 1. The second heat exchanger additionally includes a second inlet 50 adjacent its first
26 outlet 46 for receiving the first portion of the primary stream from the corresponding first
27 outlet 46 of the first heat exchanger.

28 **[0062]** To complete the cooperation between the first and second heat exchangers, the
29 second heat exchanger 26 additionally includes a third inlet 52 adjacent the first inlet 44
30 thereof for receiving the second portion of the primary stream from the corresponding second

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1 outlet 48 of the first heat exchanger 24.

2 [0063] The various inlets and outlets of the two heat exchangers permit the independent
3 operation thereof as typical counterflow heat exchangers, with the additional cooperation
4 therebetween for initially cooling the hot primary stream, splitting that initially dry-cooled
5 stream, and then adding mass or moisture thereto in the two counterflow channels of the two
6 heat exchangers. The primary stream is therefore reheated by both the primary and secondary
7 streams and discharged from the second heat exchanger with added mass by saturation for
8 improving the efficiency of operation of the exemplary turbine system.

9 [0064] Figure 5 illustrates an alternate embodiment of the duplex exchanger saturator,
10 designated 22B, in which the two heat exchangers 24,26 are disposed in a common tubular
11 shell 40, with an additional septum 40b dividing the shell into two corresponding
12 compartments for the heat exchangers. The various elements of the two heat exchangers
13 illustrated in Figure 2 may be similarly introduced in the single-shell saturator illustrated in
14 Figure 5 with corresponding inlets and outlets interconnected in the same manner as Figure 2
15 for providing the cooperating two main flow channels 28,32 and two counterflow channels
16 30,34.

17 [0065] Accordingly, the single-shell dual heat exchanger embodiment of Figure 5 is
18 equivalent in function to the dual-shell dual heat exchanger embodiment of Figure 2, both
19 represented schematically in Figure 1. It is noted that excess evaporative fluid, such as water,
20 may be injected into the first heat exchanger and will itself be cooled, with the water being
21 coldest near the discharge end of the first tubes 28.

22 [0066] The first heat exchanger, therefore, also includes one or more water outlets 54 for
23 removing the excess water, with the coldest excess water being suitably reused in the overall
24 cycle, such as being injected into the counterflow channels of the heat exchangers. The
25 second heat exchanger 26 also includes one or more water outlets 54 for removing the excess
26 evaporative fluid therefrom. The dual shell saturator configuration was previously introduced
27 in the two provisional patent applications. Features of that description are reproduced as
28 follows.

29 [0067] The dual shell and tube humidifier or saturator 22 illustrated in Figure 2 uses a
30 principal of thermodynamics dealing with cooling a gas sensibly and then evaporating a liquid

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1 into this cooled gas stream. To better describe this process consider the gas to be cooled as
2 hot compressed product air 18 and the liquid to be evaporated as water 38.

3 **[0068]** The tube humidifier contains two shell and tube heat exchangers: lower heat
4 exchanger 24 and top heat exchanger 26. The lower heat exchanger 24 comprises long and
5 short tubes 28 which are placed between a tube sheet and full baffle at opposite ends, with
6 additional partial baffles 30 in between, and an inlet 44 for the compressed product air 18.

7 **[0069]** The long tubes, whose forward ends are secured in the tube sheet and aft ends in the
8 full baffle, discharge a first portion of the product air. The short tubes, whose forward ends
9 are also secured in the tube sheet, have aft ends mounted short of the full baffle to form a blind
10 counterflow chamber for passing inside it a second portion of the compressed product air 18.
11 Finally the second portion is directed via a forward duct to the top heat exchanger 26. And,
12 the first portion of the primary airstream 18 is directed from the lower heat exchanger 24 to
13 the top heat exchanger 26 via an aft duct.

14 **[0070]** The top heat exchanger 26 comprises only long tubes 32, which are placed between
15 two tube sheets. The top exchanger further includes an inlet 44 and outlet 46 at opposite ends
16 for the stack gas 20 from the turbine, and another outlet 48 for the compressed product air 18.

17 **[0071]** All tubes 28,32 have the outside porous or grooved surface 42, which are wetted by
18 water 38. Both heat exchangers 24,26 also contain water trays, feeders, inlets, outlets, and
19 baffles for steady distribution of the primary airstreams in the open spaces between the tubes.

20 **[0072]** In the lower heat exchanger 24 the hot compressed product air 18 enters the tubes 28
21 and passes to their opposite ends being cooled sensibly. The first portion of this cooled air is
22 then passed directly via the aft duct to the top heat exchanger 26. The second portion of the
23 cooled compressed air is released into the blind counterflow chamber at the aft ends of the
24 open short tubes and directed across the tubes in counterflow.

25 **[0073]** Water 38 is distributed by a water feeder and is sprayed into the second portion of
26 air, and also wets the outside porous surface of the long and short tubes 28 in the blind
27 chamber. This causes the compressed product air to be cooled continually over the tube's
28 length by water evaporating into the second air portion. The psychometric analysis shows that
29 the temperature of the compressed product air inside the tubes 28 is reduced sensibly and
30 incrementally over the tube length towards its dew point temperature.

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1 [0074] This is accomplished by the second air portion in the blind counterflow chamber of
2 the lower heat exchanger 24 having been sensibly cooled to a lower wet bulb temperature,
3 which ideally is equal to the incoming air's dew point temperature. As the second air portion
4 continues to travel inside the blind chamber and across the wetted outside porous surfaces of
5 the tubes 28 in counterflow it is heated by the compressed product air which is passing inside
6 the tubes 28, raising the wet bulb temperature of the second air portion and ensuring that this
7 portion of air is at saturation.

8 [0075] The top heat exchanger 26 receives the two split portions of the primary airstream
9 from the lower heat exchanger, one near each end. The first portion of the dry cooled air that
10 leaves the tubes 28 from the lower heat exchanger 24 moves via the aft duct to the top heat
11 exchanger 26 and then is directed in counterflow to the hot stack gas 20 which is passing
12 inside the tubes 32 of the top heat exchanger. This first portion of the dry cooled compressed
13 air has water added causing this portion to be humidified as in the lower heat exchanger 24
14 and cooling the stack gas 20 inside the tubes 32 of the top heat exchanger 26 to the wet bulb
15 temperature of the first air portion which is passing along the wetted outside porous surface of
16 the tubes 32.

17 [0076] The wet bulb temperature of the first air portion increases as the air moves across the
18 tubes 32 absorbing the heat from the stack gas 20 inside the tubes and raising absolute
19 humidity of the air. At some predetermined point the second air portion enters via the forward
20 duct to the top heat exchanger to rejoin the first air portion. Together these first and second
21 portions of air then move in counterflow to the hot stack gas 20 from the turbine. Finally hot
22 saturated compressed product air 18 is directed via the outlet 48 to the combustion chamber.

23 [0077] It may be desirable to add heat to, or superheat the compressed product air, and at
24 that point water will not be sprayed in part of the outside porous surface of the tubes 32 of the
25 top heat exchanger preventing any further humidification, adding only heat. At the same time
26 the compressed product air is cooled inside the tubes 28 of the lower heat exchanger, the
27 excess water near the aft ends of the open short tubes is also cooled to the wet bulb
28 temperature of the air around it. Therefore the water near the aft ends of the short tubes is the
29 coolest.

30 [0078] For this reason it may be desirable to have several water inlets and outlets so that the

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1 coldest water can be used to cool for instance the intercooler of a multistage compressor of a
2 gas turbine. Additional water inlets may be desirable to help regulate the amount of
3 superheating in the top heat exchanger.

4 [0079] Compressed gas fuel or natural gas very often has a lower water vapor pressure and
5 therefore temperature than the compressed product air 18 at the same temperature. This
6 means that gas fuel also has the ability to absorb water vapor at a lower temperature. When
7 using compressed gas fuel instead of the compressed product air as the primary gas stream 18
8 in the humidifier 22, water 38 can be evaporated into the gas fuel with waste heat from the
9 stack gas exhaust 20 used to force a high evaporation rate.

10 [0080] If the dew point temperature of the compressed fuel gas is reduced below 200°F, then
11 the water vapor in the stack gas may begin to condense out depending on the humidity in the
12 stack gas. This gives added waste heat recovery in both sensible and latent heat forms of the
13 exhaust stack gas.

14 [0081] The dual shell and tube heat and mass exchanger or saturator 22 has many
15 advantages compared with known apparatuses. These include:

16 1. there is no need for after compression heat exchangers to cool the compressed air
17 with the cool water from the saturator, before entering the saturator;

18 2. there is no need for a recuperator as the function thereof is included in the top
19 exchanger;

20 3. there is no need for a humidification tower that cannot fully saturate the air and is
21 dependent on the heat exchangers;

22 4. there is no need for a boiler to better humidify the compressed air;

23 5. only the available heat limits humidification of the compressed air;

24 6. control of the humidification of the compressed air and superheating this air with
25 the exhaust gas is simply effected by the amount and location of water entering the shell side
26 of the top exchanger; the properties of high-pressure air and water vapor mixtures are not well
27 known creating problems in sizing and design of existing equipment; this is not the case with
28 the humidifier 22 as it is self-regulating by its design and can easily be adjusted to whatever
29 conditions are desired;

30 7. cooling water can be drawn from the cold-water outlet of the lower heat exchanger

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1 if desired;

2 8. there is less pressure drop, as there are fewer pieces of equipment to travel through;

3 9. less surface area is needed, as the heat transfer rate is higher due to the evaporation
4 on the outside of the tubes;

5 10. the temperature difference is greater as the tube wall temperature will become the
6 wet bulb temperature of the surrounding air;

7 11. the initial cost and on-going operating costs are less;

8 12. the concept has been well proven in several types of apparatus;

9 13. well defined and proven heat and mass transfer calculations for shell and tube heat
10 exchangers can be used for sizing; and

11 14. plate heat exchangers can also be used with this same concept.

12 [0082] Examples of the plate duplex exchangers are illustrated schematically in Figures
13 6-10. A first embodiment of a split counterflow plate saturator, designated 22C, is illustrated
14 in Figures 6 and 7. The first and second heat exchangers 24,26 described above are
15 incorporated in a common stack of thin plates 56 suitably mounted in a box housing or frame
16 58.

17 [0083] Each of the plates includes a bank of first tubes 28 integral therewith which
18 collectively define the first main flow channel of the first heat exchanger; and the first
19 counterflow channel 30 is defined between corresponding portions of the plate outside the first
20 tubes 28.

21 [0084] Similarly, each of the plates also includes a bank of second tubes 32 integral
22 therewith which collectively define the second main channel of the second heat exchanger;
23 and the second counterflow channel 34 is defined between corresponding portions of the
24 plates outside the second tubes.

25 [0085] As shown in Figure 7, the first and second tubes 28,32 are integrally formed in the
26 plates in any conventional manner and project into the corresponding counterflow channels
27 30,34 defined laterally between the stacked plates.

28 [0086] As shown in Figure 6, the first tubes 28 in each plate have a common inlet 44 at the
29 forward end of the plate, and discrete outlets 46 at opposite ends of the plates, preferably
30 terminating short thereof, for providing flow communication to both counterflow channels

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1 30,34 in common. The second tubes 32 in each of the plates also have a common inlet 44 at
2 the forward end thereof, and a common outlet 46 at the aft end of the plates.

3 [0087] In this way, the secondary fluid stream 20 is fully contained within the second main
4 channels 32 as it flows through the upper portion of the enclosure. The primary gas stream 18
5 is channeled through the first main channels 28 and discharged at the aft ends thereof into the
6 space between the stacked plates defining both counterflow channels 30,32. The cool, dry
7 primary stream discharged from the stack of first main channels 28 then flows in counterflow
8 between the plates back to the forward end of the plates for discharge through the common
9 outlet 48 in the housing.

10 [0088] The stack of plates 56 illustrated in Figures 6 and 7 may also include an optional
11 septum or seal 40c separating the banks of first and second tubes 28,32 from each other. The
12 septum also separates the first and second counterflow channels 30,34 at least in part from
13 each other between the stacked plates.

14 [0089] As shown in Figure 6, the septum splits the cool dry primary stream 18 discharged
15 from the first tubes 28 at the aft end of the septum for flow through both counterflow channels
16 30,34. The two counterflow channels rejoin at the forward end of the septum 40c for
17 discharging the rejoined, saturated primary stream through the common outlet 48.

18 [0090] The means 36 for injecting the evaporative fluid 38 into the two counterflow
19 channels 30,34 may be suitably configured for the plate configuration for injecting the fluid
20 between adjacent plates and outside the corresponding integral tubes 28,32 thereof.

21 [0091] In the various embodiments disclosed herein, the velocity of the primary gas stream
22 being channeled through the two heat exchangers will be as large as practical which permits
23 injection of the evaporative fluid at any convenient location, with the fluid being rapidly
24 disbursed in the fast moving primary stream.

25 [0092] Figures 8-10 illustrate yet another embodiment of the duplex exchanger configured
26 as a split counterflow saturator, designated 22E. In this embodiment, the two heat exchangers
27 24,26 include a stack of alternating first and second plates 60,62 suitably mounted in a frame
28 58 defined by opposite endplates joined by tie bolts. Each of the plates includes a partition or
29 seal 64 on corresponding sides thereof which collectively define the first and second main
30 flow channels 28,32 between alternating pairs of the plates.

1 [0093] Correspondingly, the first and second counterflow channels 30,34 are defined on
2 opposite sides of each of the stacked plates between different alternating pairs thereof. In this
3 way, the two dry-side main channels 28,32 are disposed between two plates, and the two
4 wet-side counterflow channels 30,34 are disposed between the next two plates, and the
5 sequence alternates from plate to plate in the stack.

6 [0094] As shown in Figure 8, each of the first plates 60 further includes an inlet aperture 44
7 at the forward end thereof for channeling the primary gas stream 18 into the first main channel
8 28. The first plates also include an outlet or transfer aperture 46 at the opposite aft end thereof
9 for discharging the primary stream from the first main channel 28. Since the outlet aperture
10 46 extends through the plate it provides direct flow communication to the first and second
11 counterflow channels 32,34 disposed on the back side of the corresponding main channels
12 28,32.

13 [0095] The first plates also include yet another inlet 44 at the forward ends thereof for
14 introducing the secondary fluid stream 20 into the second main channel 32. And, another
15 outlet aperture 46 is disposed in the aft end of the first plates for discharging the secondary
16 stream from the second main channel.

17 [0096] Correspondingly, each of the second plates 62 illustrated in Figure 8 includes a
18 respective outlet aperture 48 at the forward end thereof for providing a common outlet for the
19 two counterflow channels 30,34.

20 [0097] As shown in Figures 8 and 10, the two plates 60,62 further include complementary
21 herringbone corrugations 66 which adjoin each other, and surrounding perimeter seals 68 to
22 further define the first and second main flow channels 28,32 between corresponding sides of
23 the plates, and also defining the cooperating first and second counterflow channels 30,34 on
24 the opposite sides of the plates, with the main channels and counterflow channels alternating
25 between the stacked plates.

26 [0098] Corrugated plate heat exchangers are conventional and may be suitably modified and
27 configured for introducing the cooperating two heat exchangers in the manner illustrated
28 schematically in Figure 1. The corrugations 66 are locally formed in the otherwise thin, flat
29 plates 60,62 to define the different heat exchangers 24,26 bounded within flat lands. The
30 partition and perimeter seals 64,68 are located at the flat lands to seal the boundaries of the

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1 heat exchangers.

2 [0099] As shown in Figures 8-10, the primary airstream 18 is suitably channeled in parallel
3 through the first main channel 28 between alternating plates and is discharged through the aft
4 outlet apertures 46 thereof into the spaces between the next plates. Those spaces include the
5 two counterflow channels 30,34 through which the primary stream is channeled back toward
6 the forward end of the plates.

7 [0100] Correspondingly, the hot secondary stream is channeled in parallel through the
8 second main channel 32 between the stacked plates and is discharged through the aft outlet
9 aperture 46.

10 [0101] The coolant and saturant injecting means 36 are suitably introduced in the stacked
11 plate configuration for injecting a common evaporative fluid 38 into the first and second
12 counterflow channels 30,34 downstream of the aft outlets 46 in the first plates 60 where the
13 outlets join the first main channel 28 with both the first and second counterflow channels
14 30,34. In this way, the evaporative fluid is injected into both counterflow channels for
15 providing evaporative cooling of the primary and secondary streams in the two main channels
16 28,32.

17 [0102] If desired, a suitably short septum seal 40e as illustrated in Figure 8 may be disposed
18 at least in part between the stacked first and second plates 60,62 opposite to the partition seal
19 64 to separate the corresponding first and second counterflow channels 30,34. In this way, the
20 primary gas stream is split at the common entrance ends of the two counterflow channels for
21 separate counterflow therealong and then rejoins at the forward ends of the plates prior to
22 discharge from the common outlet aperture 48 in the several plates.

23 [0103] The herringbone corrugations 66 illustrated in Figures 8-10 are the same on both
24 sides of each plate, but have different and complementary configurations for the two plate
25 types 60,62. In this way, they can be stacked together and sealed to define the different
26 flowpaths therein; with the flow streams following serpentine paths between the overlapping
27 corrugations where they abut or adjoin.

28 [0104] The corresponding inlets and outlets for the plates are defined by the respective
29 apertures therein and the interplate seals therebetween, and the interruptions therein in a
30 conventional manner. The first plates 60 have four active apertures for flow control, and the

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1 second plates 62 have two active apertures for flow control; with only the aft outlet 46 of the
2 first main channel 28 being common to the stacked plates to internally feed the primary stream
3 18 to the two counterflow channels 30,34 formed between alternating plates.

4 [0105] Accordingly, the two plates 60,62 collectively have five active apertures between
5 them, which are identically located in the entire stack of plates in corresponding alignment for
6 providing continuous inlet, outlet, and transfer flowpaths or manifolds therethrough.

7 [0106] In this way, two inlet conduits may be provided in one endplate 58 as shown in
8 Figures 8-10 for isolated feeding of the primary and secondary inlet streams 18,20 to only the
9 first and second main channels 28,32 of each plate. Suitable interplate seals confine the inlet
10 flows, and prevent inlet flow of these streams to the first and second counterflow channels
11 30,34.

12 [0107] The aft outlet apertures 46 of the first main channels 28 also define common inlets to
13 the first and second counterflow channels 30,34, and those transfer apertures are aligned
14 through the stack of plates and bound by the perimeter seals 68 to contain the flow internal to
15 the plate stack.

16 [0108] And, two outlet conduits may be provided in the opposite endplate 58 for isolated
17 discharge of the primary and secondary outlet streams from the corresponding second
18 counterflow channels 34 and second main channels 32 as confined by suitable interplate seals.

19 [0109] As indicated above, Figure 1 illustrates a simple schematic representation of the split
20 counterflow saturator form of the evaporative duplex counterheat exchanger and the
21 cooperating two heat exchangers therein. The primary gas stream is sensibly cooled in the
22 main flow channel by evaporation of the fluid in the cooperating counterflow channel. The
23 cool and dry primary stream is then split for counterflow back through the first heat exchanger
24 as well as through the second heat exchanger. The hot secondary stream is channeled through
25 the main channel of the second heat exchanger for evaporating the fluid in the corresponding
26 counterflow channel to humidity or saturate the previously cooled and dry primary stream.
27 The primary stream is then discharged from the saturator for any suitable purpose, such as
28 adding mass to the combustion process in the exemplary turbine system illustrated.

29 [0110] Figures 2-10 illustrate various forms of evaporative duplex counterheat exchangers
30 specifically cooperating in the manner illustrated in Figure 1. Any type of conventional heat

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1 exchanger including tube-in-shell and plate designs may be configured and modified for use in
2 the manner illustrated in Figure 1. Figures 2-10 are merely exemplary of the various forms of
3 otherwise conventional heat exchangers which may be used for achieving the improved split
4 counterflow configuration disclosed above.

5 [0111] The split counterflow saturator may be configured for any other applications in
6 which it is desired to saturate one fluid in another. And, the evaporative duplex counterheat
7 exchanger may be configured and used in other applications for advantageous use of the
8 saturated primary stream, or the cooled secondary stream, or the cooled condensate liquid of
9 the evaporative fluid.

10 [0112] While there have been described herein what are considered to be preferred and
11 exemplary embodiments of the present invention, other modifications of the invention shall be
12 apparent to those skilled in the art from the teachings herein, and it is, therefore, desired to be
13 secured in the appended claims all such modifications as fall within the true spirit and scope of
14 the invention.

15 [0113] Accordingly, what is desired to be secured by Letters Patent of the United States is
16 the invention as defined and differentiated in the following claims in which we claim: